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9 Technical ~~Memorandum~~

6 ROTARY EXPANDER ENGINE DEVELOPMENT PROGRAM

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ABSTRACT

This paper describes some results and the status of a program to develop a 250 hp (186 kW) rotary expander engine for use with an external combustion source or in a closed-cycle power system. In the first phase of the program, parts from a Curtiss-Wright RC1-60 internal combustion engine were combined with a new housing, end plates and a set of rotary valves to produce the first rotary expander, the RE1-60. The RE1-60 was tested at a maximum power output of 155 hp (116 kW) at 4800 rpm when supplied with combustion gas at 1000°F (538°C) and 600 psia (4.14 MPa). Details of the RE1-60 and test results are presented.

The second rotary expander, the RE1-11, was designed with the expander's kinematics similar to that of Wankel's earlier engines; the main shaft/eccentric is stationary while the rotor and housing rotate. Power take-off occurs from the housing. Compared to the RE1-60 this kinematic change eliminates two balance weights and a fly wheel, and permits location of the valves closer to the expander's working chamber. Details of the RE1-11 design are given in the paper along with solutions to some challenging casting and fabrication problems. The current status of the RE1-11 fabrication program and plans for half-power testing are summarized.

ADMINISTRATIVE INFORMATION

This memorandum was prepared under Project No. C30504, "Underwater Propulsion Block Program:" Principal Investigator T.J.Black, Code 3642. The sponsoring activity is Naval Material Command (MAT-08T2) Mr. I. Jaffe.

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INTRODUCTION

THE ROTARY PISTON OR WANKEL EXPANDER ENGINE is a particular member of a class of rotary piston machines (1)*. Although major interest in the Wankel engine has been in the gasoline-air internal combustion engine application, the rotary piston machine is a candidate for open- and closed-cycle use as a positive-displacement expander since its high specific power values, either on a weight or volume basis, are attractive. In addition a dynamically balanced, pure rotary machine should generate lower and less directional vibrations than alternative reciprocating machines.

The overall objective of the program is to develop a 250 hp (186 kW) rotary expander engine for use with an external combustion source or in a closed-cycle power system. The objective of the first phase of the program was to verify the concept of the rotary piston machine as an expander and to develop experimentally verified analytical techniques for design of the 250 hp expander. This design, together with fabrication and testing, would be completed in the program's second phase. A redesigned Curtiss-Wright Corporation RC1-60(2) internal combustion engine was used to create the first phase rotary expander engine, the RE1-60. A newly designed RE1-11 rotary expander will be used in the second phase of the program. As with the designation of the RC engine series, the integer after the RE code represents the number of rotors while the value following the hyphen represents the displacement volume, in cubic inches, associated with one face of the rotor.

RE1-60 PROGRAM RESULTS

EXPANDER DETAILS - It was apparent that the program could be expedited and costs reduced if some of the components of an existing Wankel internal combustion engine were employed. The RE1-60, used extensively by the Curtiss-Wright Corporation (2) was utilized under a lease agreement. Figure 1 shows the basic components of the RE1-60 rotary piston expander. The rotor, seals, bearing, mainshaft, and balance weights were used without modification.

* Numbers in parentheses designate references at end of paper.

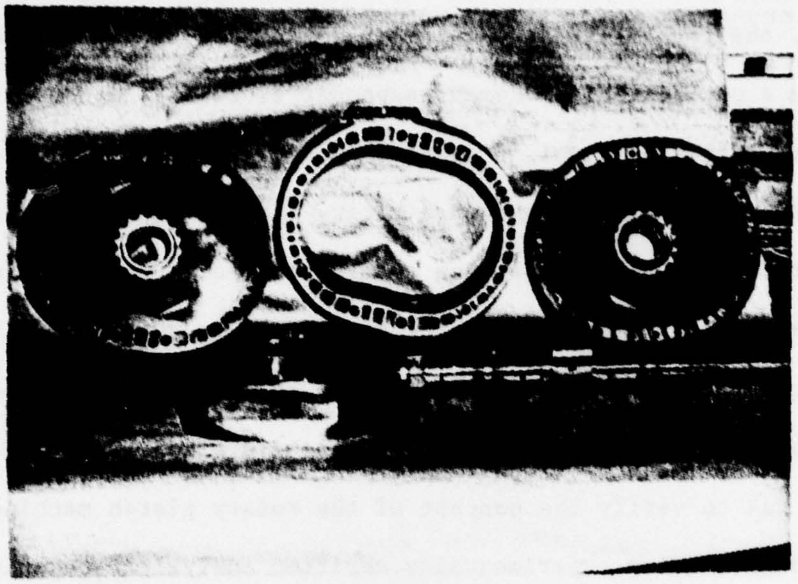


Figure 1 - Basic RE1-60 Rotary Piston Expander Components

The covers and flywheel were modified. The rotor housing and the aft and forward housings were redesigned, and two rotary valves added.

The engine housings incorporated a "one-pass" water coolant system, which simplified the casting design and manufacture. The original housings were aligned together and constrained by four dowels. The expander design incorporated four additional dowels to reduce the mechanical stress in the rotor housing to the level of the internal combustion engine.

Two large bosses, located diametrically opposite on the outer periphery of the rotor housing, serve as the interface seats for the rotary valves. The engine has two inlet ports because the expander concept easily adapts itself to two power cycles per output shaft revolution compared to one for the RC1-60. Dual exhaust ports designed into the end housings provide the necessary area to exhaust the working gas from the engine.

The rotary valves were designed with two ports in the valve spool feeding one rotor housing inlet port. This reduced the valve size, pressure-balanced the spool and decreased the valve angular velocity. The gas intake valve flow area schedule was fixed by the physical size and speed of the rotary valve. The minimum spool clearance was designed for the maximum conditions of 700 psia (4830 kPa) and 1000°F (538°C). Testing at lower conditions resulted in larger clearances and more blowby in the valve. Criteria on rotary valve materials were based largely on temperature expansion coefficients and availability of material.

Table 1 summarizes some geometry data for the RE1-60. The source for each value is indicated in the table. Ansdale (3) contains a description of the Wankel nomenclature. The first five quantities are determined by the RC1-60 engine. The expander displacement volume is double the value for one rotor face since two pairs of inlet and exhaust ports were employed (figure 1). The theoretical minimum volume was calculated from:

Table 1 - RE1-60 Expander Geometry Data

QUANTITY	SYMBOL	SOURCE	VALUE	
Generating Radius	R	RC1-60	13.14 cm	5.175 in
Eccentricity	e	RC1-60	1.905 cm	0.750 in
Housing Displacement	a	RC1-60	0.102 cm	0.040 in
Rotor Width	W	RC1-60	7.62 cm	3.000 in
Rotor Depression Volume	V_r	RC1-60	57.35 cu cm	3.500 cu in
$\overline{K}$ Ratio	$\overline{K}$	R/e	6.900	6.900
K Ratio	K	$(R+a)/e$	6.953	6.953
Displacement Volume Per Rotor Face	V_d	$3\sqrt{3}\,eW(R+a)$	999.1 cu cm	60.97 cu in
Expander Displacement Volume	D_v	$2 \cdot V_d$	1998.2 cu cm	121.94 cu in
Theoretical Minimum Volume	V'_{min}	Equation (1)	57.7 cu cm	3.52 cu in
Housing Displacement Volume	V_a	aWL_{tr}	18.5 cu cm	1.13 cu in
Valve Volume	V_v	Estimated	16.4 cu cm	1.00 cu in
Minimum Volume Per Rotor Face	V_{min}	$V'_{min}+V_r+V_a+V_v$	150 cu cm	9.15 cu in
Maximum Volume Per Rotor Face	V_{max}	$V_{min}+V_d$	1149 cu cm	70.12 cu in
Clearance Ratio	C_L	V_{min}/V_d	0.150	0.150
Compression Ratio	ε	V_{max}/V_{min}	7.66	7.66
Volume At Inlet Port Closing	V_{co}	Rotary Valve Geometry	394.3 cu cm	24.06 cu in
Dimensionless V_{co}	$\overline{V}_{co}$	V_{co}/V_d	0.395	0.395
Crankshaft Angle For V_{co}	θ_{sco}	Equation (2)	88.92°	88.92°
Volume At Exhaust Port Closing	V_{ex}	Exhaust Port Location	374.9 cu cm	22.88 cu in
Dimensionless V_{ex}	$\overline{V}_{ex}$	V_{ex}/V_d	0.375	0.375
Crankshaft Angle For V_{ex}	θ_{sex}	Equation (2)	455°	455°

$$V'_{min} = e^2 W \left\{ \frac{\pi}{3} + 2\sqrt{K^2-9} + 2\left(\frac{K^2}{9} + 2\right) \sin^{-1}\left(\frac{3}{K}\right) \right\}$$

$$- \frac{1}{2} 3\sqrt{3}\, K e^2 W \quad . \qquad (1)$$

The housing displacement, a, is the amount that the theoretical epitrochoidal housing is displaced parallel to itself, and is equal to the apex seal tip radius, to minimize relative motion between the tip seal and rotor. V_a is the volume associated with this displacement. The valve volume was estimated from drawings and is associated with the passageway between the inner housing surface and the rotary valve surface. The actual minimum volume for each rotor face is the sum of the four volumes indicated. The volumes at inlet and exhaust port closings are indicated together with the corresponding crankshaft angles measured from the rotor in the minimum volume position. The equation relating volume and crank angle is:

$$V = V_{min} + \frac{V_d}{2}\left\{1 - \cos\left(\frac{2}{3}\theta_s\right)\right\} \quad . \qquad (2)$$

From figure 1 it will be seen that an exhaust port is located in both the aft and forward housing plates. The inlet valve opened at TDC ($\theta_s = 0^o$) and the exhaust port was uncovered at BDC ($\theta_s = 270^o$).

EXPERIMENTAL EVALUATION - The RE1-60 was tested using gases generated by an air-water-ethyl alcohol combustor and a water brake dynamometer. The rotary valves were driven by timing belts from the engine output shaft. Extensive pressure, temperature and mass flow rate instrumentation was employed, including four Kistler piezoelectric pressure transducers located in the expander housing wall. Complete details of the experimental evaluation are presented in reference (4).

Table 2 summarizes the experimental program. Before each hot gas run the expander was spun to about 3000 rpm with air alone to insure proper operation of the expander and auxiliary equipment. A 22 minute endurance run was made at 55hp (41 kW) and 3230 rpm with 450 psia (31 kPa) gas at temperatures between 590°F(310°C) and 830°F (443°C). A maximum power run was made on 11/8/74 with maximum power of 153 hp (114 kW) at 4700 rpm achieved with hot gas at 1300 psia (8960 kPa) and 1170°F (632°C).

ANALYTICAL RESULTS - Three levels of analysis were used in this study. At the first level, efficiencies, specific expendables consumption, heat balances, etc. were calculated. Detailed results are given in reference (4). At the second level, the results of DiPippo (5) were used to determine the card factor C_F and the flow coefficient C_W. The card factor is the ratio of the actual area of the pV diagram to the area of a theoretical pV diagram (5). Similarly, the flow coefficient is a ratio of the actual massflow rate through the expander to the ideal flow rate for the ideal pV diagram. The use of DiPippo's analysis requires data on the inlet pressure and temperature, the exhaust pressure, engine geometry, and specific heat ratio k and molecular weight M. Values of k and M were obtained from a NASA chemical equilibrium computer program (6). At the third level of the analysis, Bowlus (4) has developed a computer simulation of positive-displacement expanders including the Wankel type. During the inlet and exhaust processes the control volume equations for conservation of mass and the first law of thermodynamics are written with instantaneous mass flow rates used from known valve or port flow area schedules as a function of crank angle. Discharge coefficients C_D for the valves are required. During the compression and expansion processes the system equations are written. For all processes heat transfer to the wall is included through the use of a heat-transfer coefficient and the instantaneous wall area. An equation of state and thermodynamic properties for the working fluid are required. One version of the program uses actual data for steam while the version used for this study uses perfect gas approximations based on the NASA calculations. This third

Table 2 - Hot Gas Run Summary

RUN-DATE	DURATION SECONDS	COMBUSTION CONDITIONS PRESSURE (kPa)	TEMPERATURE (°C)	SPEED RPM	POWER KW
1-8/19/74	70	1380-3280	199-246	2160-2916	14-22
2-9/12/74	90	2760-5380	349-618	3000-3852	34-67
3-9/18/74	220	621-3480	174-335	850-3000	2-32
4-9/25/74	80	3140-5620	288-610	2850-4000	28-74
5-10/18/74	125	1000-3280	177-382	1450-2950	5-29
6-10/24/74	310	2960	399	3350	43
7-10/29/74	180	3240	382	3140	39
8-11/6/74	1325	3100	310-443	3230	41
9-11/8/74	270	3070-8960	243-632	2670-4700	21-114

level of analysis permits the calculation of the pV diagram (and indicated power), the mass flow rates, etc. if the discharge coefficients and heat-transfer coefficient are known, or the deduction of these coefficients from the data for later design use.

EXPERIMENTAL RESULTS - Figures 2 and 3 show measured and computed pV diagrams for an air spin and a hot-gas run, respectively. Using discharge coefficients for the inlet and exhaust ports of 0.8 and 0.6, respectively, and a heat transfer coefficient of about 600 Btu/hr F-ft^2 (3.41 kJ/S-$^{\circ}$C-m^2) produces the agreement shown. The pV diagram data yield the indicated horsepower and hence the mechanical efficiency η_m of the engine, defined as the ratio of the brake to indicated horsepower. For this study, the power to operate the valves was added to the brake power after experimentally determining the valve power. Typical valves of η_m were about 0.7.

Figure 4 shows the gross horsepower divided by the valve inlet pressure versus engine speed for all 44 data sets. The DiPippo analysis suggests that these parameters should have a linear relationship if the $C_F \eta_m$ product is constant for all conditions. The data indicate that for this first model of the Wankel expander values of C_F were about 0.4. Reference (4) contains a summary of all data and additional results.

CONCLUSIONS - The concept of a rotary expander engine was completely verified during the test program summarized in table 2. The expander started easily on both air and combustion gases. No quantitative vibrational data were obtained on this expander, but it was apparent that the expander performed with the smoothness expected from rotary engines. No problems were encountered in the cooling, lubrication, or sealing of the expander. Likewise, structural integrity and durability were demonstrated. For the RE1-60, values of important performance parameters were: η_m = 0.7, C_F = 0.4, C_W = 0.7, thermal efficiency between 0.3 and 0.6, C_D = 0.8 (inlet valve), C_D = 0.6 (exhaust valve), heat

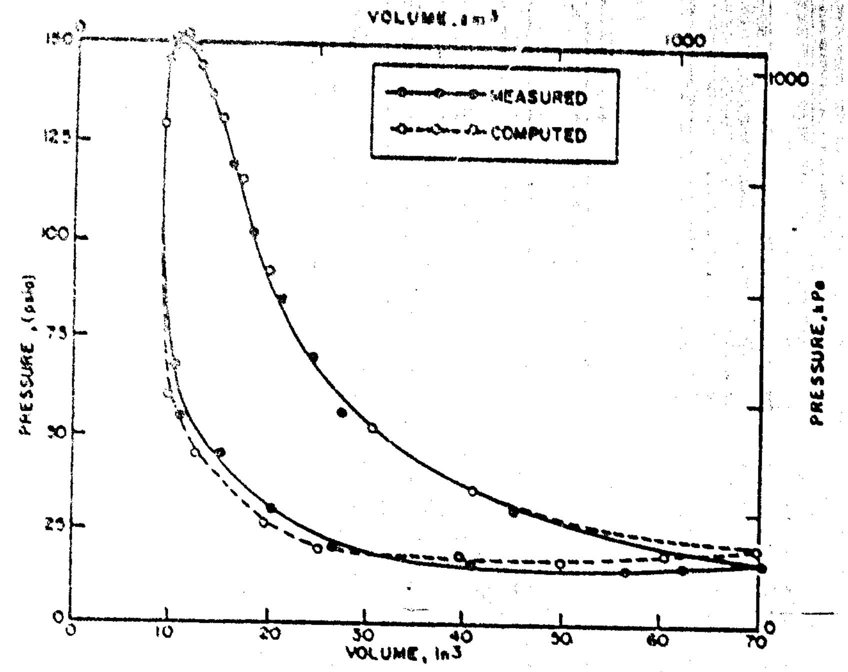


Figure 2 - Air Spin Data

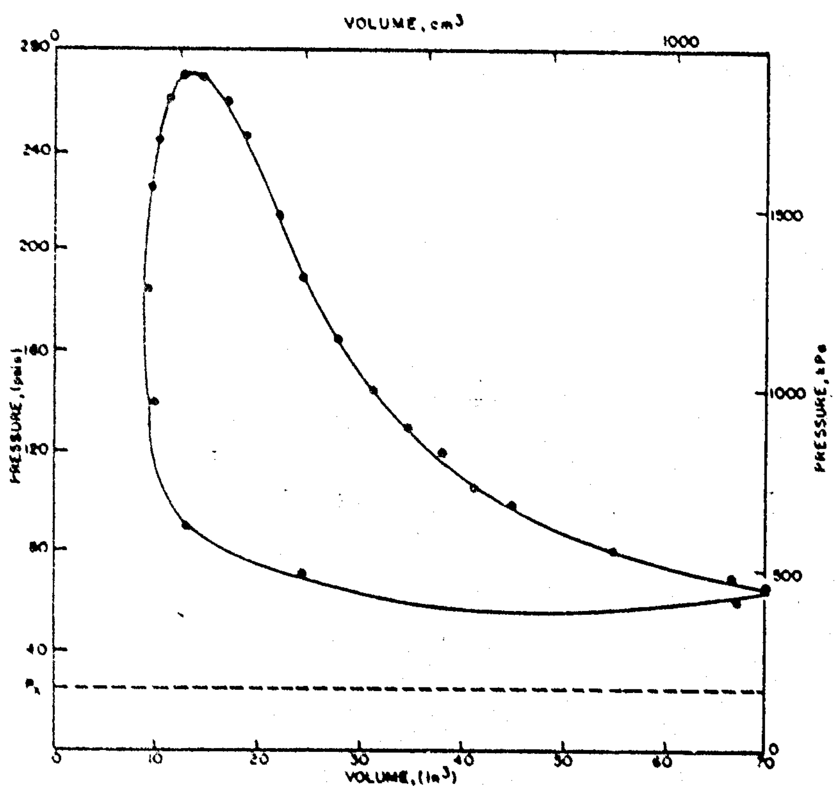


Figure 3 - Hot-Gas Run Data (Run No. 8, Test No. 36)

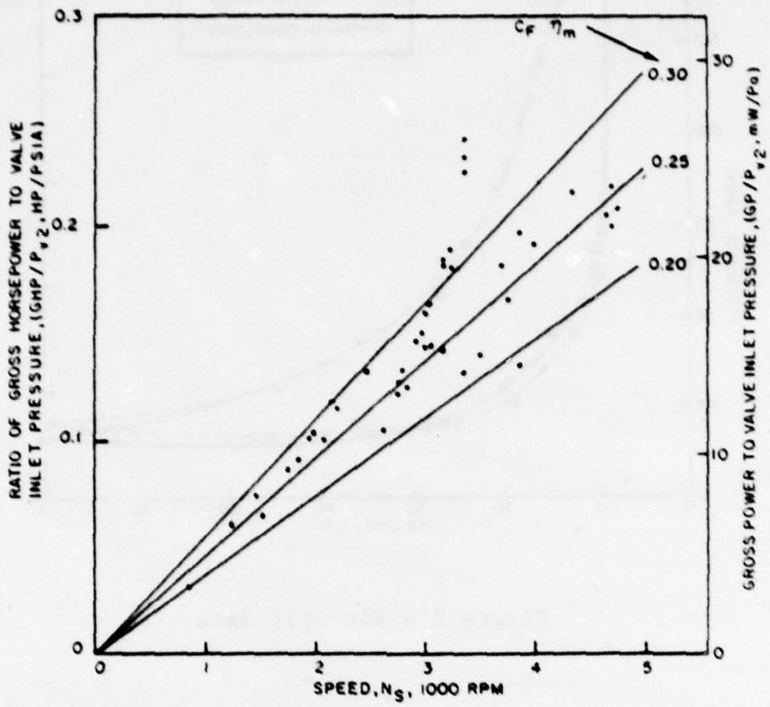


Figure 4 - Variation of GHP/PV2 with Expander Speed

transfer coefficent to housing surfaces was about 600 Btu/hr-F-ft^2 (3.41 kJ/s-^{o}C-m^2). The primary mechanical deficiency of this first expander was the valve system. The filling process was poor because of the inlet valve opening schedule, line dynamics associated with having the combustion chamber far from the inlet valves, too small an inlet area and leakage in the valve. The exhaust process could be improved by opening before BDC to ensure adequate exhausting.

RE1-11 PROGRAM RESULTS

KINEMATIC ARRANGEMENT - Figure 5 shows schematically the kinematic arrangements for the RE1-60 and RE1-11. In figure 5A the rotor, housing and eccentric sections are shown in their minimum volume (TDC) position which is also the position for $\theta_s = 0$. The points noted by A and B are the geometric centers of the housing and rotor, respectively. Figure 5B shows a general position of the rotor for the RE1-60. Since the eccentric section is also the output shaft section a rotation of the output shaft by θ_s produces the same rotation of the eccentric section. The radius ratio between the rotor gear and the stationary housing gear is 3:2, and produces a rotation of the rotor through the angle $\frac{1}{3}\,\theta_s$. These relationships are also indicated in table 3. It should be noted that the definition of the rotor angle corresponds to the counterclockwise direction which introduces the negative sign noted in table 3 for θ_r. This notation has been retained since it is the notation used by Ansdale (3). In summary, for the RE1-60 the housing is stationary, and the eccentric - output shaft section rotates three times faster than the rotor. Figure 5C shows a general rotational position for the RE1-11. Since the housing and the output shaft section are bolted together, rotation of the output shaft by an angle θ_s causes the same rotation in the housing. For the RE1-11 the gear radius ratio is still 3:2 and, consequently, the rotor rotates through the angle $\frac{2}{3}\,\theta_s$. The eccentric section is now stationary. The rotor rotates in the same

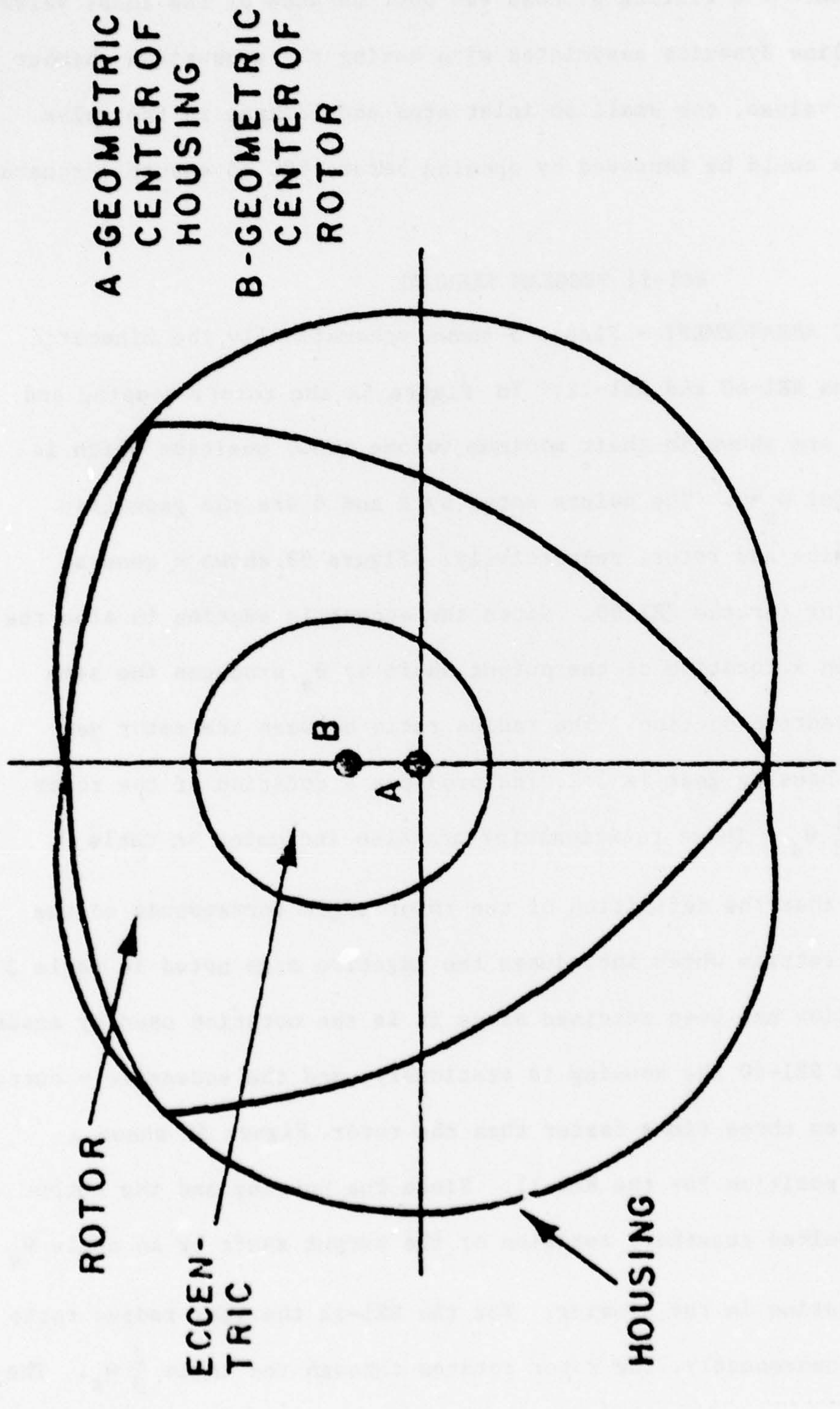


Figure 5A. - Minimum Volume Position for Components

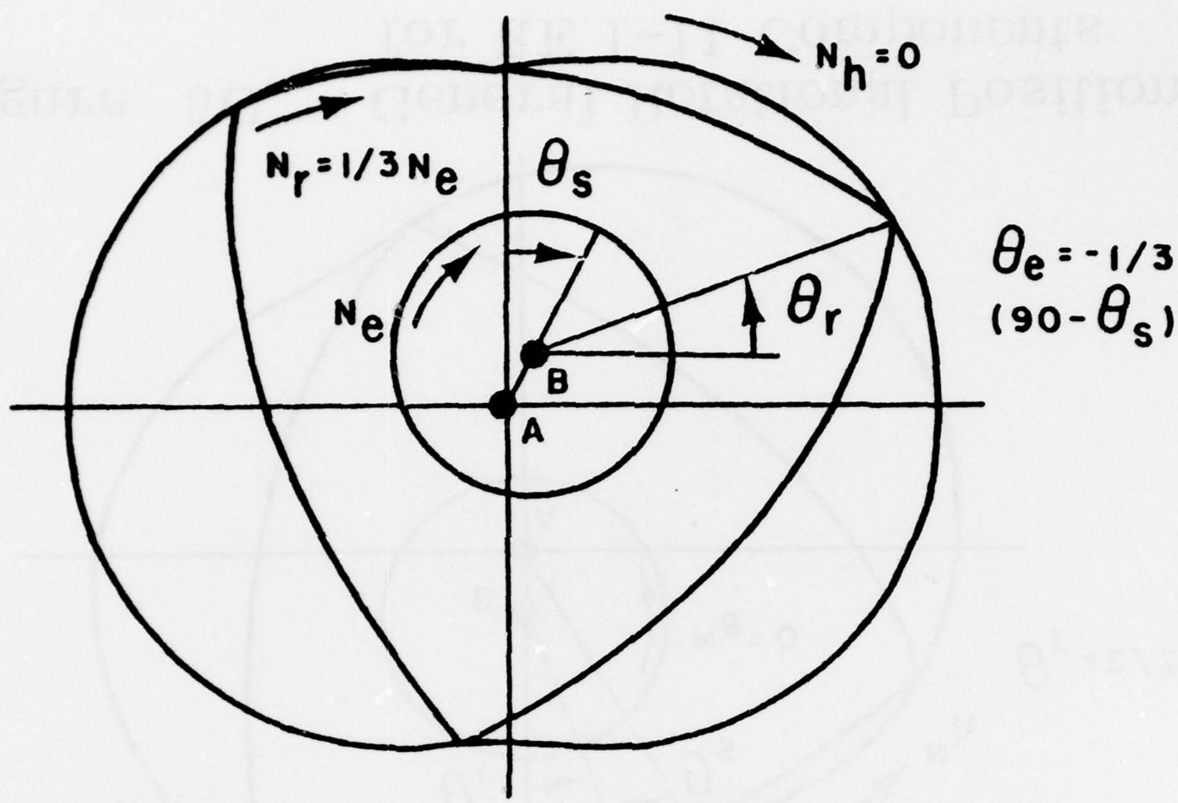


Figure 5B. - General Rotational Position for REI -60 Components

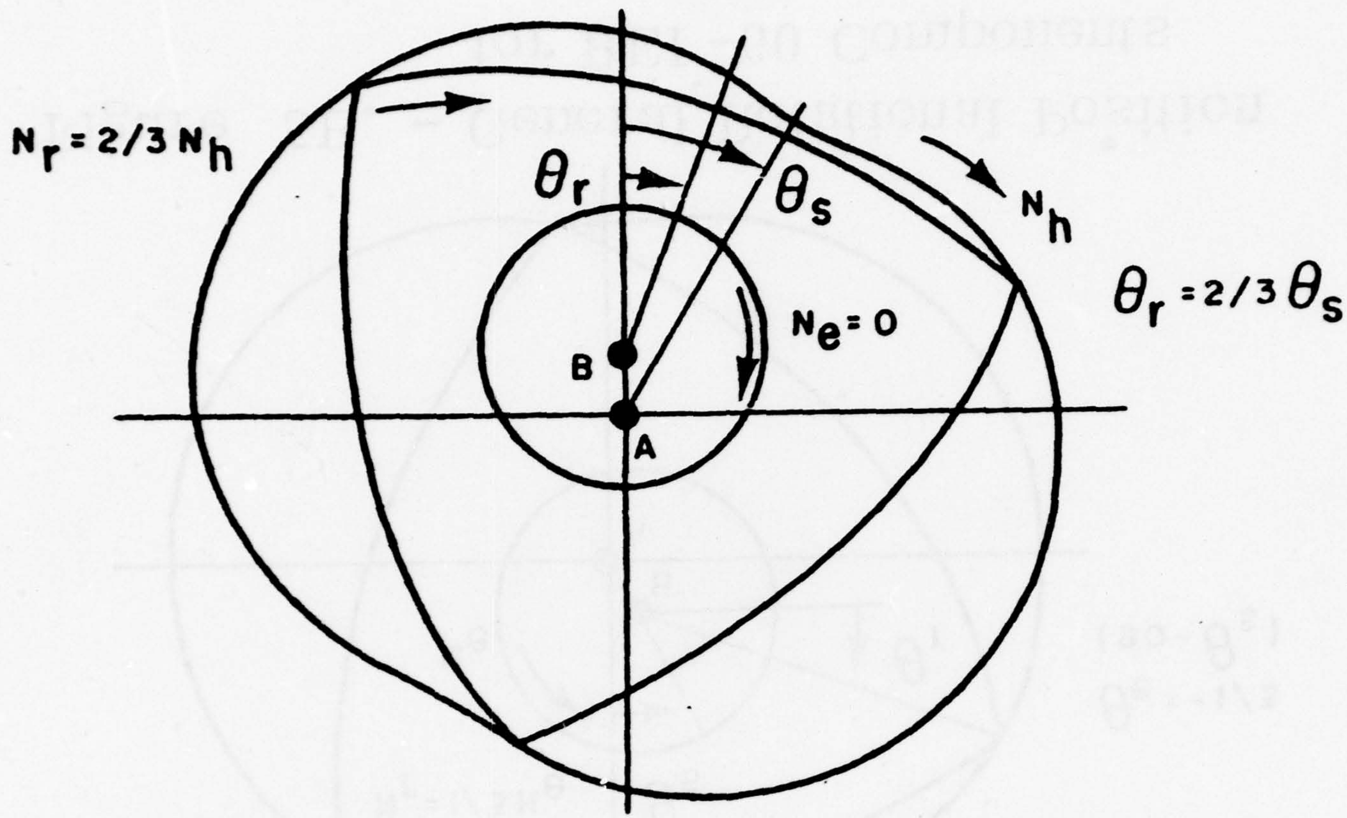


Figure 5C. - General Rotational Position for RE 1-11 Components

Table 3 - Angular Displacement and Speed Relationships for the RE1-60 and RE1-11 Expanders

	RE1-60	RE1-11
ANGULAR DISPLACEMENTS [1]		
COMPONENT		
ECCENTRIC, θ_e	θ_s	0
ROTOR, θ_r	$-\frac{1}{3}(90 - \theta_s)$	$\frac{2}{3}\theta_s$
HOUSING, θ_h	0	θ_s
ANGULAR SPEEDS [2]		
COMPONENT		
ECCENTRIC, N_e	N_s	0
ROTOR, N_r	$\frac{1}{3}N_s$	$\frac{2}{3}N_s$
HOUSING, N_h	0	N_s

[1] All angles measured from reference with components in minimum volume positions (see Fig. 5A). Figures 5B and 5C define positive θ directions. θ_s is rotation of expander output shaft.

[2] N_s is rotational speed of expander output shaft and is positive in positive θ_s direction.

direction as the housing but at 2/3 of the housing speed.

DESIGN DETAILS - Figure 6 shows schematically the RE1-11 expander. The overall length and outside diameter of the expander are approximately 15.5 inches (39.4 cm) and 9.75 inches (24.8 cm), respectively.

From a rotational viewpoint there are three types of elements in the expander. The rotating chamber elements are those components which are rigidly connected to the output shaft section of the expander. The second class of elements contains only the rotor. Both of these types are shown as shaded areas in figure 6. The third class of elements are those which are stationary and such elements have not been shaded in figure 6. The class of rotating chamber elements includes the exhaust cover, forward cover, rotor housing, aft cover, and output shaft section. It is clear that the forward and aft cover and rotor housing correspond to the forward and aft housings and the epitrochoidal housing, respectively, for the normal Wankel internal combustion engine. The rotating chamber elements are supported by the main bearings B and the outboard bushings and bearings C. The main bearings are mounted on the eccentric section A which also contains the rotor bearing F on which the rotor rotates. The rotating chamber elements rotate about the geometric center which lies on the main center line of the expander while the rotor rotates about its geometric center which is displaced from the expander center line. Since all rotating elements are rotating about their geometric centers there are no significant unbalanced radial forces to produce vibration and noise. This design feature also eliminates the need for the counterweights which were used in the RE1-60. The large rotary inertia of the rotating chamber elements eliminates the need for a flywheel in this design.

A stationary combustion chamber has been used which contains the stationary valve seat G at its aft end. The rotating valve seat H rotates with the exhaust cover. The relative motion of the two portions of the valve seat produces the inlet port flow area variation during cyclic operation. A circular pipe in valve seat G supplies a combustion gas stream through a slot in valve seat H and to a convergent

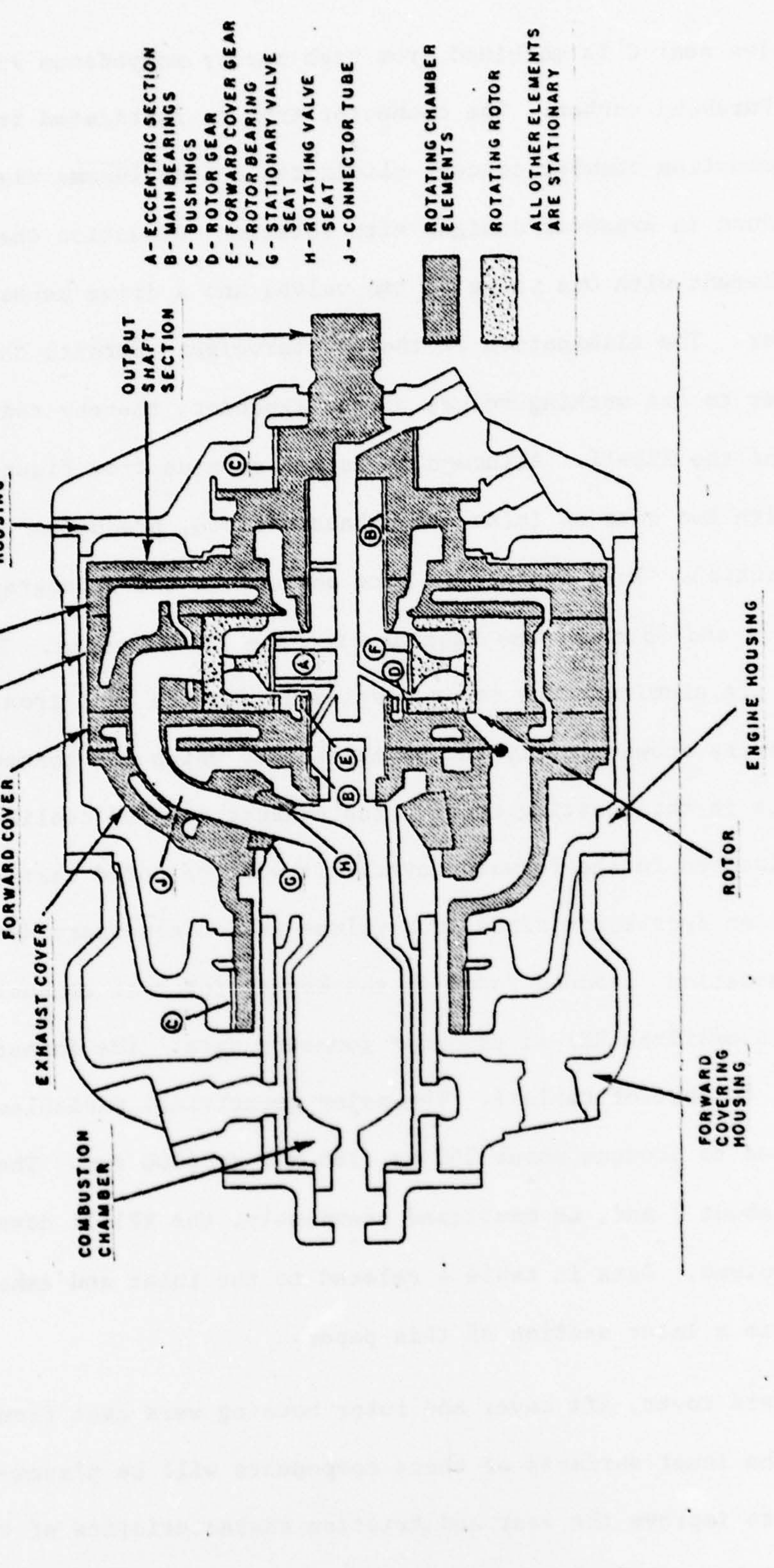


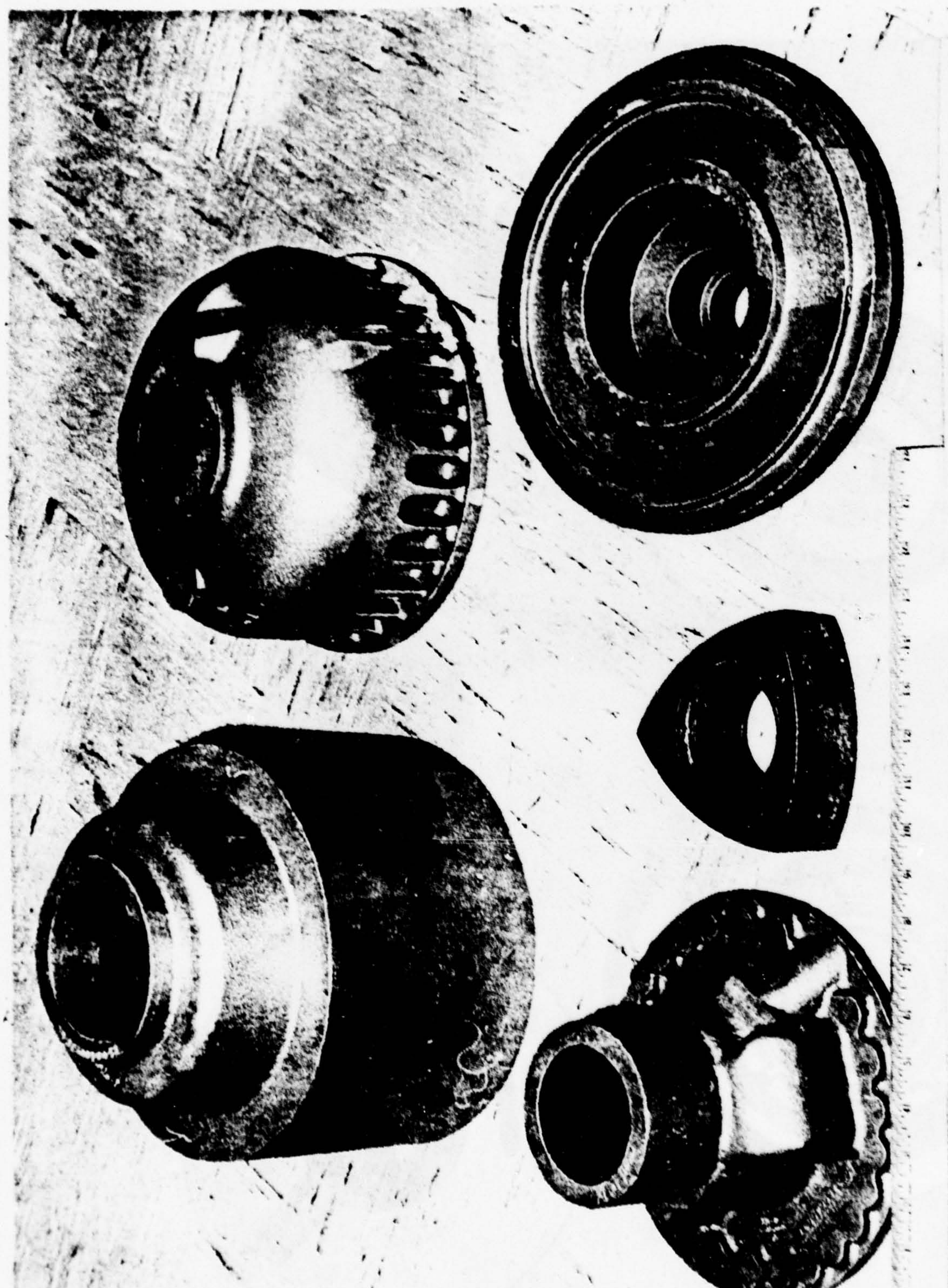
Figure 6 - Major Components of the RE1-11 Expander

cavity, directing the flow to a connector tube J, which connects to the working volume of the expander. Valve seat G is machined from high purity molybdenum and valve seat H is machined form Purebond carbon. The connector tube is fabricated from stainless steel. The stationary combustion chamber concept eliminates a troublesome high pressure propellant seal found in expander designs with rotating combustion chambers. It also permits the replacement with one valve of two valves and a drive mechanism used in the RE1-60 expander. The elimination of the counterweights permits the location of the inlet valves closer to the working volume of the expander, thereby reducing the clearance volume of the RE1-11. Although it is not obvious from figure 6, the RE1-11 operates with two sets of inlet and exhaust valves, producing two power cycles per output shaft revolution. This feature is more obvious in the photographs which follow.

Figures 7A and 7B show some castings for the RE1-11 parts. These castings, except the rotor, are aluminum; the rotor casting is ductile cast iron. The casting for the exhaust covers shows clearly the cavities into which the connector tubes fit. The two slots in this casting receive the exhaust gas and cooling water flowa through passages located in the forward cover. It will be noted that the rotor for this expander has no rotor depression since such volume is not necessary due to the absence of the combustion process found in the Wankel internal combustion engine.

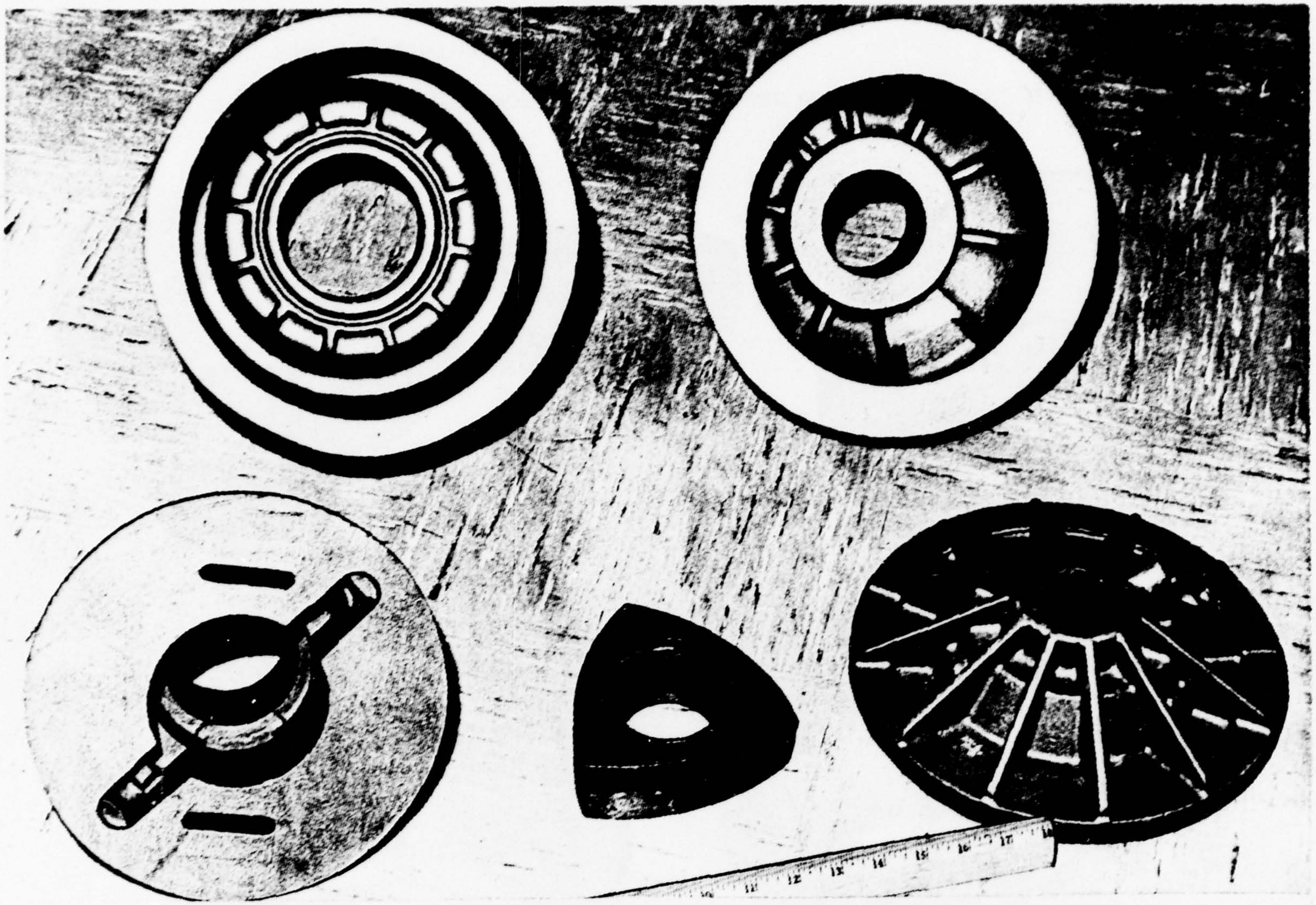
Table 4 summarizes RE1-11 expander geometry data. THe format for this table is identical to that of table 1. The major geometrical variables for the RE1-11 were selected to produce about 250 hp (186 kW) at 6000 rpm. The K ratio was maintained at about 7 and, as mentioned previously, the RE1-11 does not have a rotor depression volume. Data in table 4 related to the inlet and exhaust port operation will be discussed in a later section of this paper.

The forward cover, aft cover and rotor housing were cast from Hastealloy C. After machining, the inner surfaces of these components will be plasma-spray coated with Ferro-tic CM to improve the wear and friction characteristics of the surfaces. The selection of materials and fabrication tolerances and techniques was based on the



CLOCKWISE FROM UPPER LEFT: ENGINE HOUSING, FORWARD COVERING HOUSING, AFT COVERING HOUSING, ROTOR, EXHAUST COVER.

Figure 7A - Castings of some RE1-11 Components - Outside View



CLOCKWISE FROM UPPER LEFT: ENGINE HOUSING, FORWARD COVERING HOUSING, AFT COVERING HOUSING, ROTOR, EXHAUST COVER.

Figure 7B - Castings of some RE1-11 Components - Inside View

Table 4 - RE1-11 Expander Geometry Data

QUANTITY	SYMBOL	SOURCE	VALUE	
Generating Radius	R	Design Choice	7.450 cm	2.933 in
Eccentricity	e	Design Choice	1.08 cm	0.425 in
Housing Displacement	a	Design Choice	0.076 cm	0.030 in
Rotor Width	W	Design Choice	4.39 cm	1.730 in
Rotor Depression Volume	V_r	Design Choice	0.0 cu cm	0.0 cu in
$\bar{K}$ Ratio	$\bar{K}$	R/e	6.900	6.900
K Ratio	K	$(R+a)/e$	6.97	6.97
Displacement Volume Per Rotor Face	V_d	$3\sqrt{3}\ eW(R+a)$	185.2 cu cm	11.3 cu in
Expander Displacement Volume	D_v	$2 \cdot V_d$	370.3 cu cm	22.6 cu in
Theoretical Minimum Volume	V'_{min}	Equation (1)	10.7 cu cm	0.654 cu in
Housing Displacement Volume	V_a	aWL_{tr}	4.64 cu cm	0.283 cu in
Valve Volume	V_v	Estimated	14.5 cu cm	0.883 cu in
Minimum Volume Per Rotor Face	V_{min}	$V'_{min}+V_r+V_a+V_v$	29.8 cu cm	1.82 cu in
Maximum Volume Per Rotor Face	V_{max}	$V_{min}+V_d$	215 cu cm	13.12 cu in
Clearance Ratio	C_L	V_{min}/V_d	0.160	0.160
Compression Ratio	ε	V_{max}/V_{min}	7.20	7.20

Table 4 - RE1-11 Expander Geometry Data (Cont'd)

QUANTITY	SYMBOL	SOURCE	VALUE	
Volume at Inlet Port Opening	V_{io}	Inlet Port Location	31.23 cu cm	1.906 cu in
Dimensionless V_{io}	$\bar{V}_{io}$	V_{io}/V_d	0.168	0.168
Crankshaft Angle for V_{io}	θ_{sio}	Equation (2)	-15^o	-15^o
Volume at Inlet Port Closing	V_{co}	Inlet Port Location	76.12 cu cm	4.645 cu in
Dimensionless V_{co}	$\bar{V}_{co}$	V_{co}/V_d	0.411	0.411
Crankshaft Angle for V_{co}	θ_{sco}	Equation (2)	90.0^o	90.0^o
Volume at Exhaust Port Opening	V_{eo}	Exhaust Port Location	204.4 cu cm	12.78 cu in
Dimensionless V_{eo}	$\bar{V}_{eo}$	V_{eo}/V_d	1.131	1.131
Crankshaft Angle for V_{eo}	θ_{seo}	Equation (2)	240^o	240^o
Volume at Exhaust Port Closing	V_{ex}	Exhaust Port Location	51.49 cu cm	3.142 cu in
Dimensionless V_{ex}	$\bar{V}_{ex}$	V_{ex}/V_d	0.278	0.278
Crankshaft Angle for V_{ex}	θ_{sex}	Equation (2)	480^o	480^o

goal of successful operation of the experimental RE1-11 expander.

At this point in the program no attempt has been made to value engineer the entire expander. The design represents the concepts of the design staff of the Naval Underwater Systems Center and inputs from contracted studies by the Curtiss-Wright Corporation.

MAJOR FLOW CIRCUITS - Figure 8 shows the major flow circuits for the RE1-11. The gaseous combustion products exit the combustion chamber through the valve seat G and H to the connector tube which transfers the gas through the forward cover and rotor housing regions into the working volume of the expander. The relative motion of parts G and H produce the inlet port flow area schedule. The exhaust gases exit from the working volume through the exhaust port K into an exhaust passage located in the forward cover and exhaust cover. At point L the cooling water leaves a hole in the forward cover and flows into the same exhaust passage in the exhaust cover. Mixing of the cooling water and exhaust gas stream produces a reduction in the exhaust gas temperature. The flow downstream from L is therefore a two-phase flow of water droplets, water vapor and exhaust gases. The gases exit from the exhaust cover into an annular exhaust chamber which has been incorporated into the expander design to significantly reduce pressure fluctuations prior to discharge of the exhaust gases from the expander.

Cooling water enters the RE1-11 through the forward end of the expander and flows into a collecting ring before distribution to cooling passages surrounding the combustion chamber. The total cooling water flow is then divided into approximately equal parts to cool the inlet valve and two connector tubes. The cooling water flows through a passage in the exhaust cover which surrounds the connector tube. Upon reaching the forward cover the cooling water flows radially inward, is turned about 180° in the plane of the covered and indexed circumferentially, and then flows radially outward along the new radial flow path. It then crosses through passages in the rotor housing to a set of radial flow passages in the aft cover which are similar to those just described for the forward cover. After a set of radially inward and outward flows in the aft cover, the water recrosses the rotor housing to start a second traverse of the forward cover - rotor housing- aft cover - rotor housing flow path sequence.

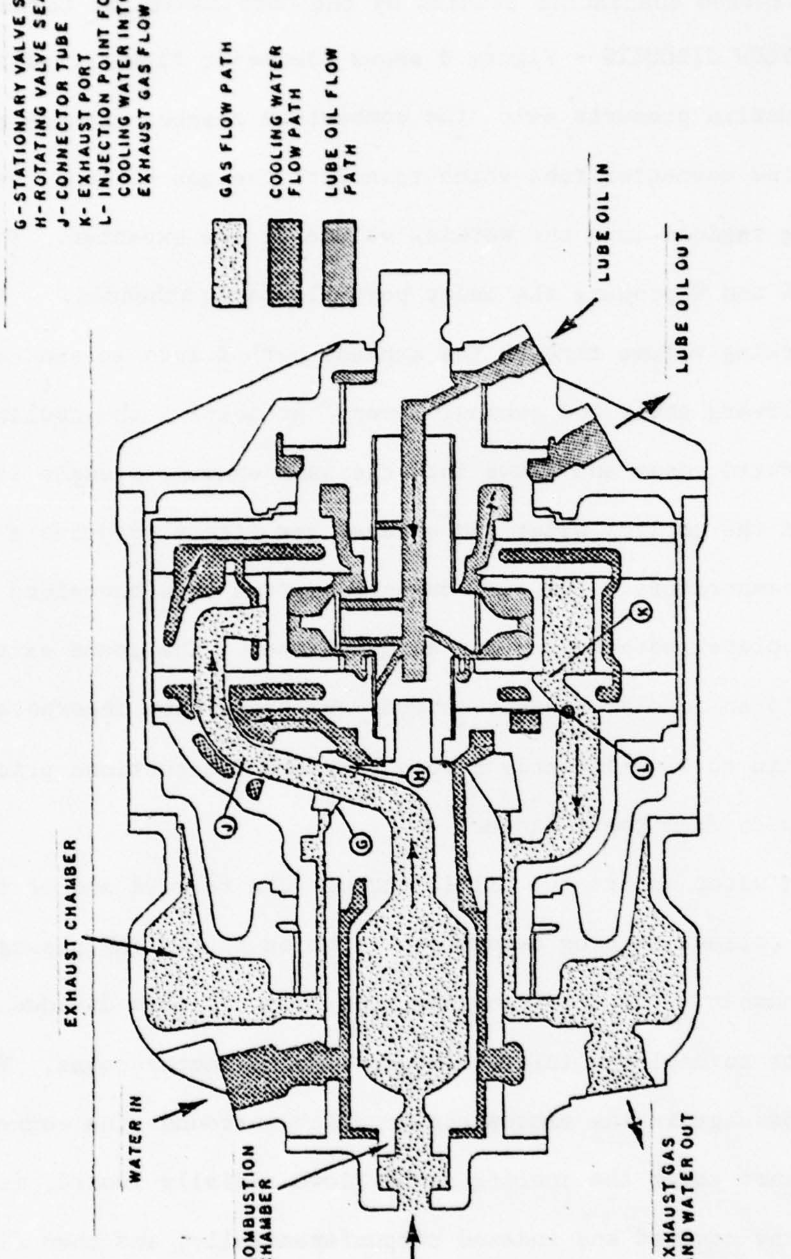


Figure 8 - Major Flow Circuits of the RE1-11 Expander

The sequence is repeated three times with the cooling water finally reaching point L where it is injected with the exhaust gas flow. A secondary flow path exists whereby cooling water flows directly from the connector tube region in the forward cover, through passages in the rotor housing and into a cooling water cavity in the aft cover. The cavity is also connected into the main cooling water circuit in the aft cover. Especially at start-up this secondary flow circuit permits rapid distribution of water to that region of the rotor housing through which the hot combustion gases are flowing and to the aft cover cooling water system.

The lubrication oil is introduced at the aft end of the RE1-11 and provides lubrication for the aft bushing. The main oil flow passes to the center line of the eccentric section where it is then distributed to the main journal bearings for rotating chamber elements and to the journal bearing for the rotor. In addition, it is distributed to the cavity inside the rotor where it provides lubrication for the forward cover and rotor gears and cooling of the internal surface of the rotor. The lubricating oil distribution system is deliberately designed to utilize the rotor motion as the driving force for the expulsion of the oil from the inner cavity of the rotor to a collecting ring from which the oil flows from the expander.

INLET AND EXHAUST PORT FLOW AREAS - Figure 9 shows a dimensionless representation of the inlet port flow area versus dimensionless crank angle position. The flow area is normalized using the maximum flow area available during the inlet process. The crank angle has been made dimensionless based on the total number of degrees of crank angle during which the inlet port is open. Figure 9 shows the inlet port flow schedules for both expanders. The schedule for the RE1-60 has a symmetrical, triangular distribution with the inlet port opening at TDC and closing at $\theta_s = 89^o$. The inlet port for the RE1-11 opens at $\theta_s = -15^o$ and closes at $\theta_s = 90^o$. It is clear that the inlet port for the RE1-11 opens sooner and remains wide open much longer than the inlet port for the RE1-60. These design changes were those suggested by the

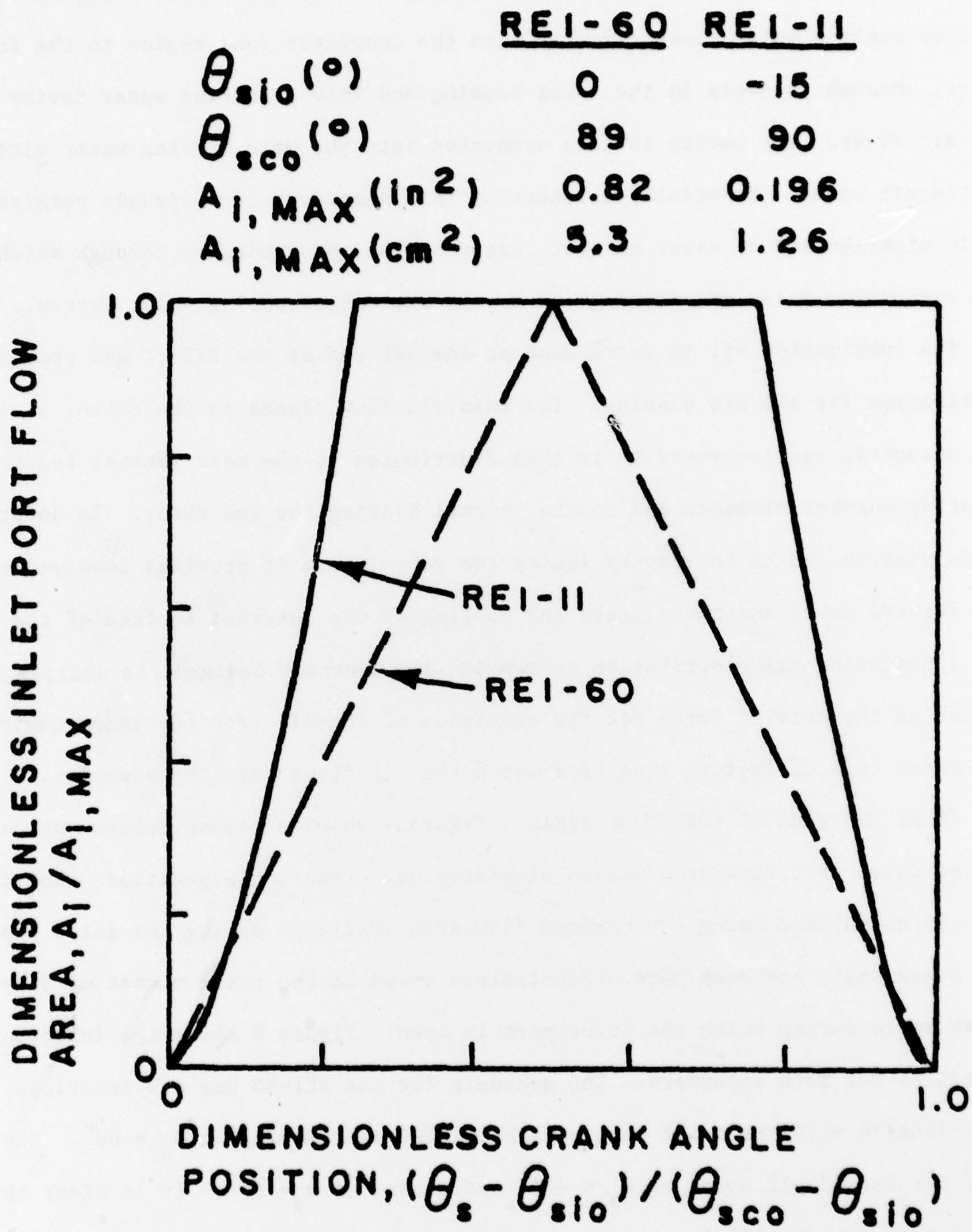


Figure 9 - Comparison of Inlet Port Flow Area Schedules

results of the RE1-60 investigation.

Figure 10 shows the exhaust port flow area schedules for the two expanders. Both expanders have similarly shaped schdules. However, the exhaust port for the RE1-11 opens approximately 40° sooner and remains open 25° longer than that of the RE1-60. In addition, changes have been made in the physical size of the exhaust port for the Rel-11 to improve the exhaust process. As suggested in the RE1-60 results, the RE1-11 exhausts only through the forward cover.

FABRICATION DETAILS - The most challenging fabrication problem was encountered in the production of the castings for the forward and aft covers and rotor housing. The castings were investment cast using the lost wax process. Figure 11 shows the dies which were used to produce the ceramic cores and wax molds, often referred to as "waxes," for the rotor housing. The final ceramic cores and waxes for the rotor housing are shown assembled in the lower center section of figure 11, in which the white parts are the ceramic cores and the dark parts are the waxes. In the final casting, the dark wax regions will be metal and the white ceramic cores will be open volumes for cooling water passages, exhaust ports, etc. Figure 12 shows the cores and waxes which are produced by the dies. The final wax model for the rotor housing is actually assembled from the two halves shown in figure 12. Figure 13 shows a partially assembled final wax model for the rotor housing with the ceramic cores placed in their proper positions. The two sides of the final wax model for the rotor housing are shown in figure 14. The ceramic core which protrudes through the inner surface of the wax model represents the extension of the connector tube through which the combustion gases will enter the working volume of the expander. The finger-like or fence-like ceramic cores represent the passages through which cooling water will pass back and forth through the rotor housing between the forward and aft covers. The two large white protrusions in the left hand protion of figure 14 are passages through which cooling water reaches regions around the connector tube in the rotor housing, The white circular cylinders shown in the right hand side of figure 14 will form the extensions of the connector tube in the rotor housing through which combustion gases pass from the front cover to the expander's working volume.

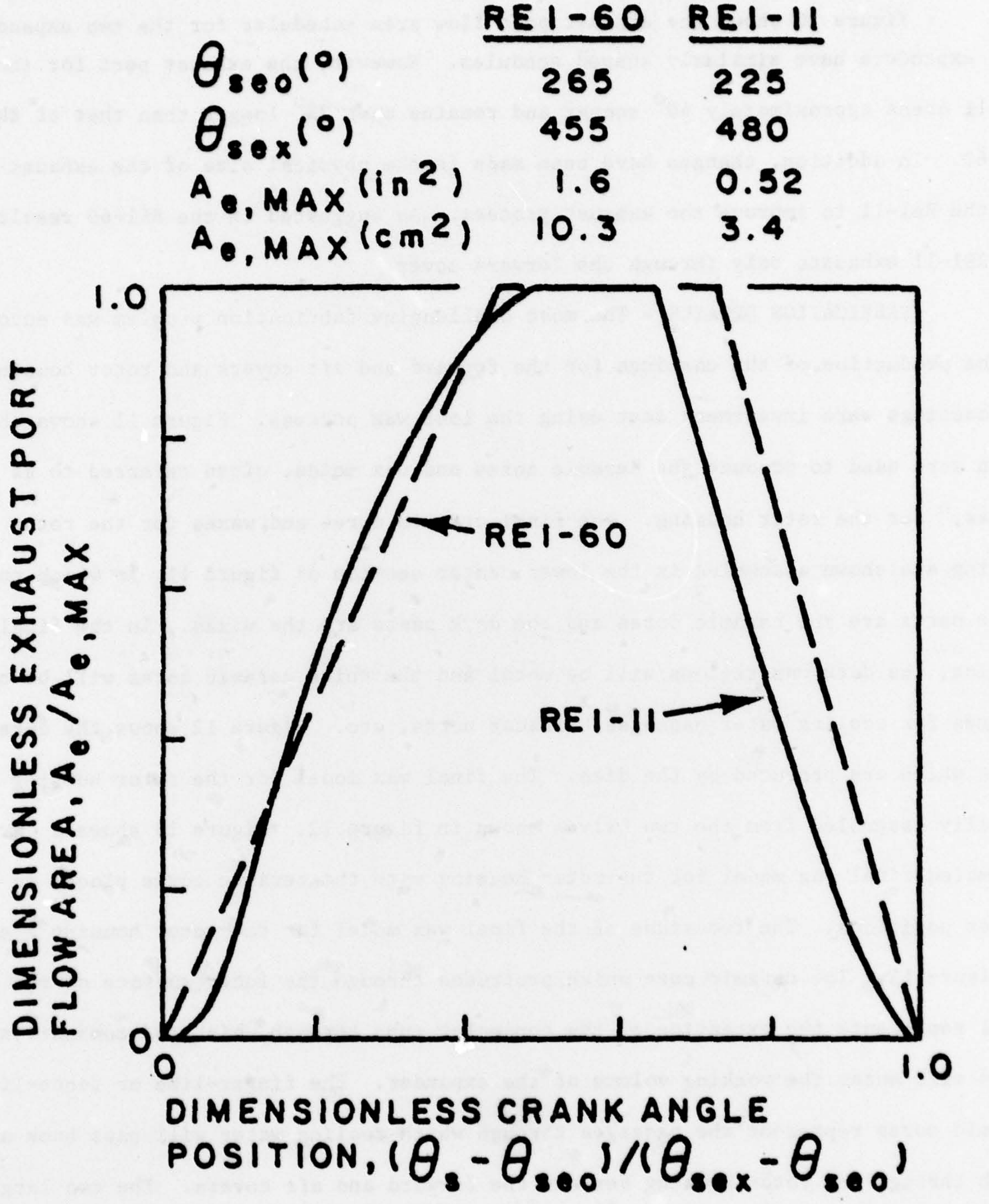


Figure 10 - Comparison of Exhaust Port Flow Area Schedules

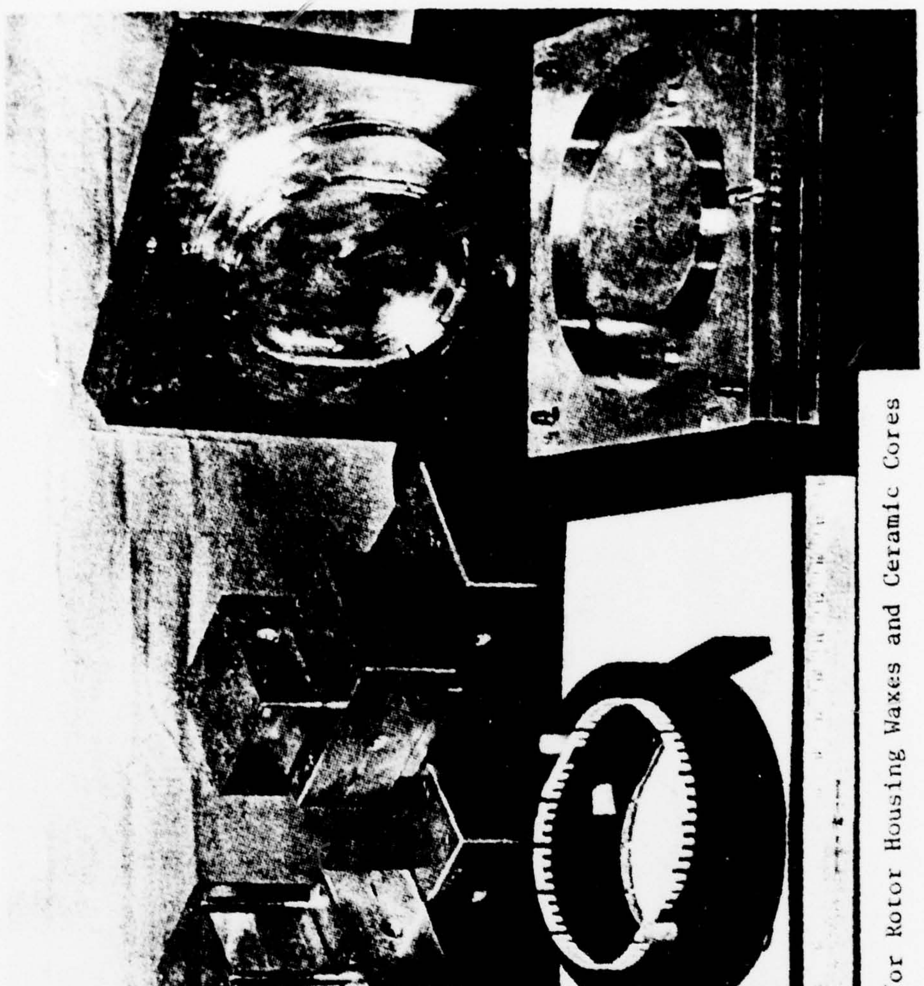


Figure 11 - Dies for Rotor Housing Waxes and Ceramic Cores

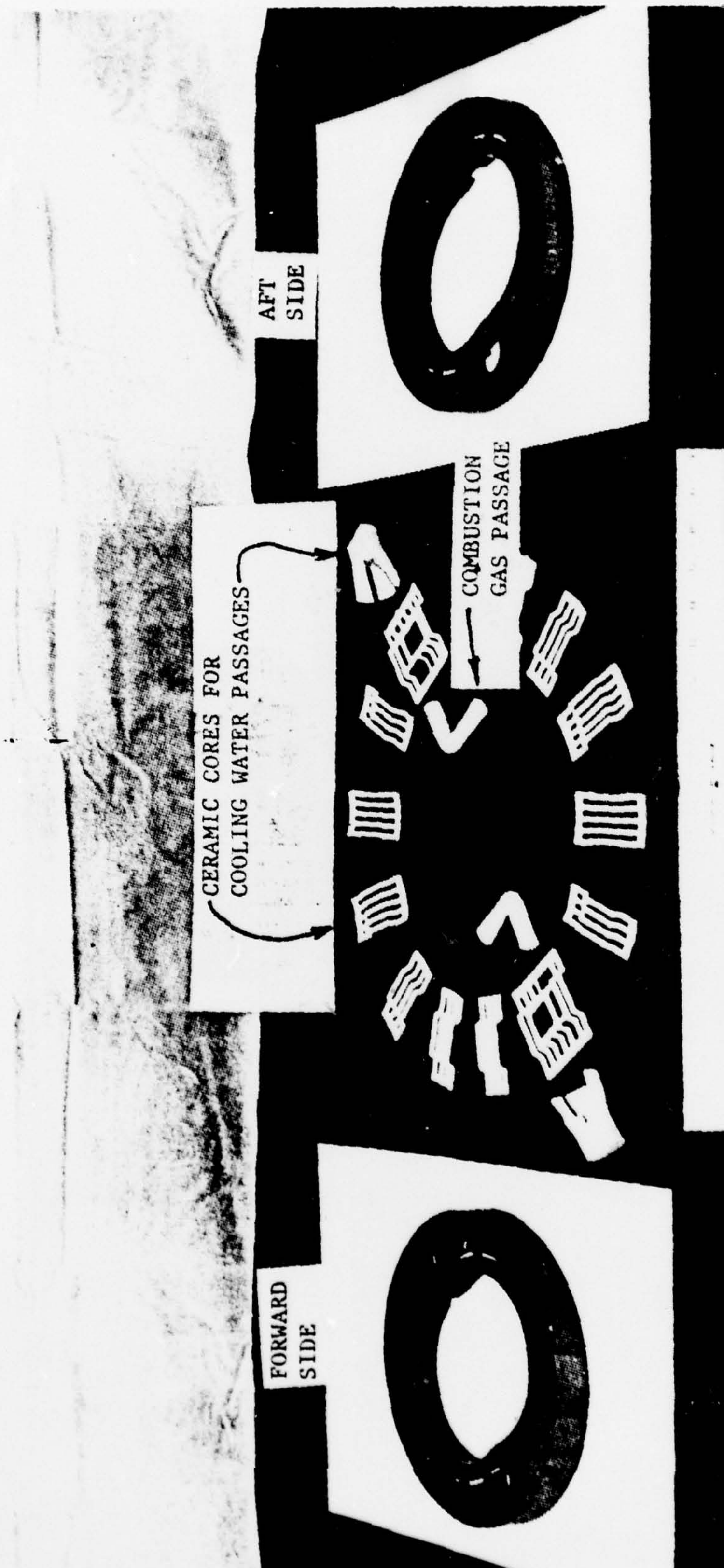


Figure 12 - Rotor Housing Waxes and Ceramic Cores

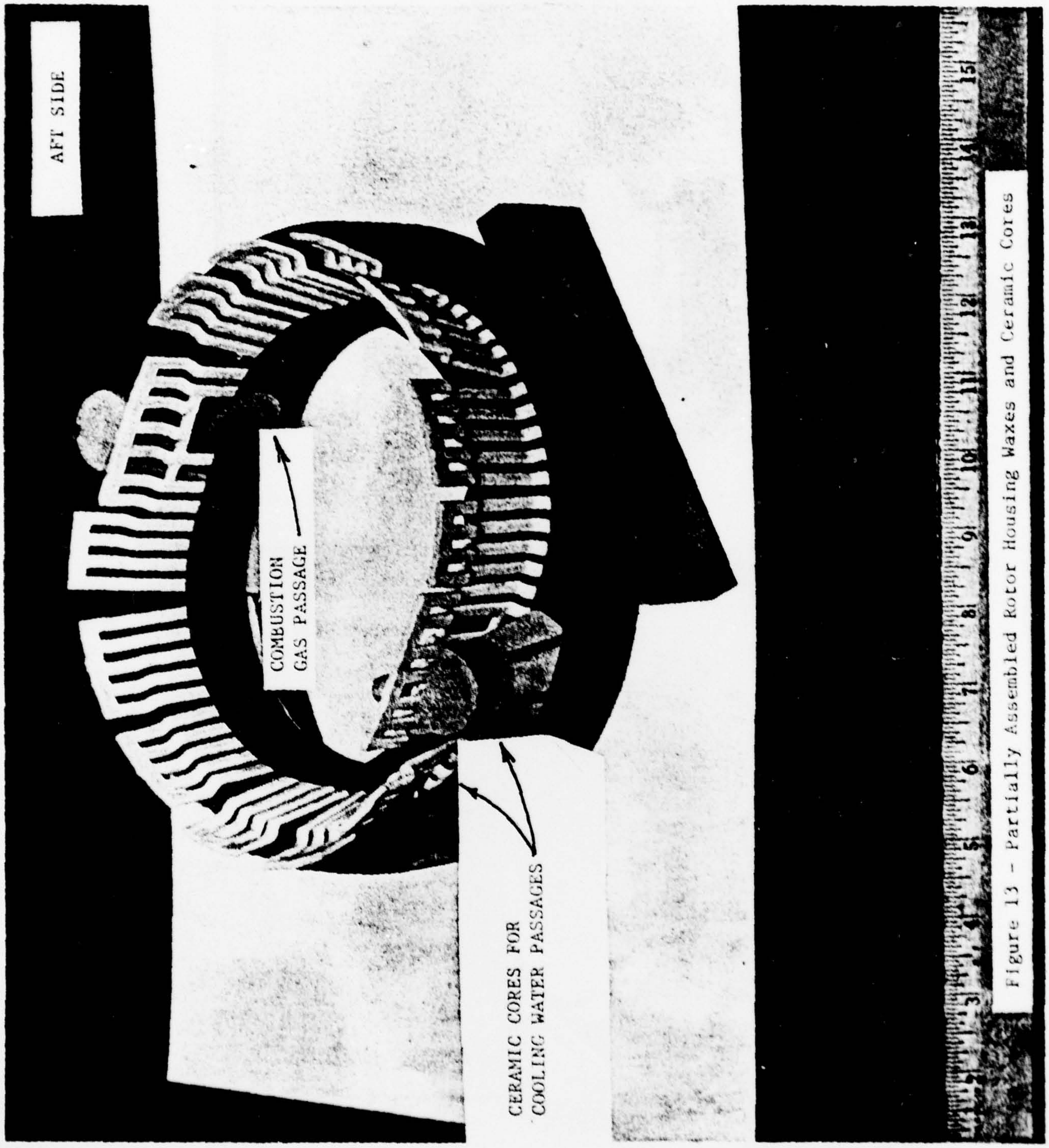


Figure 13 - Partially Assembled Rotor Housing Waxes and Ceramic Cores

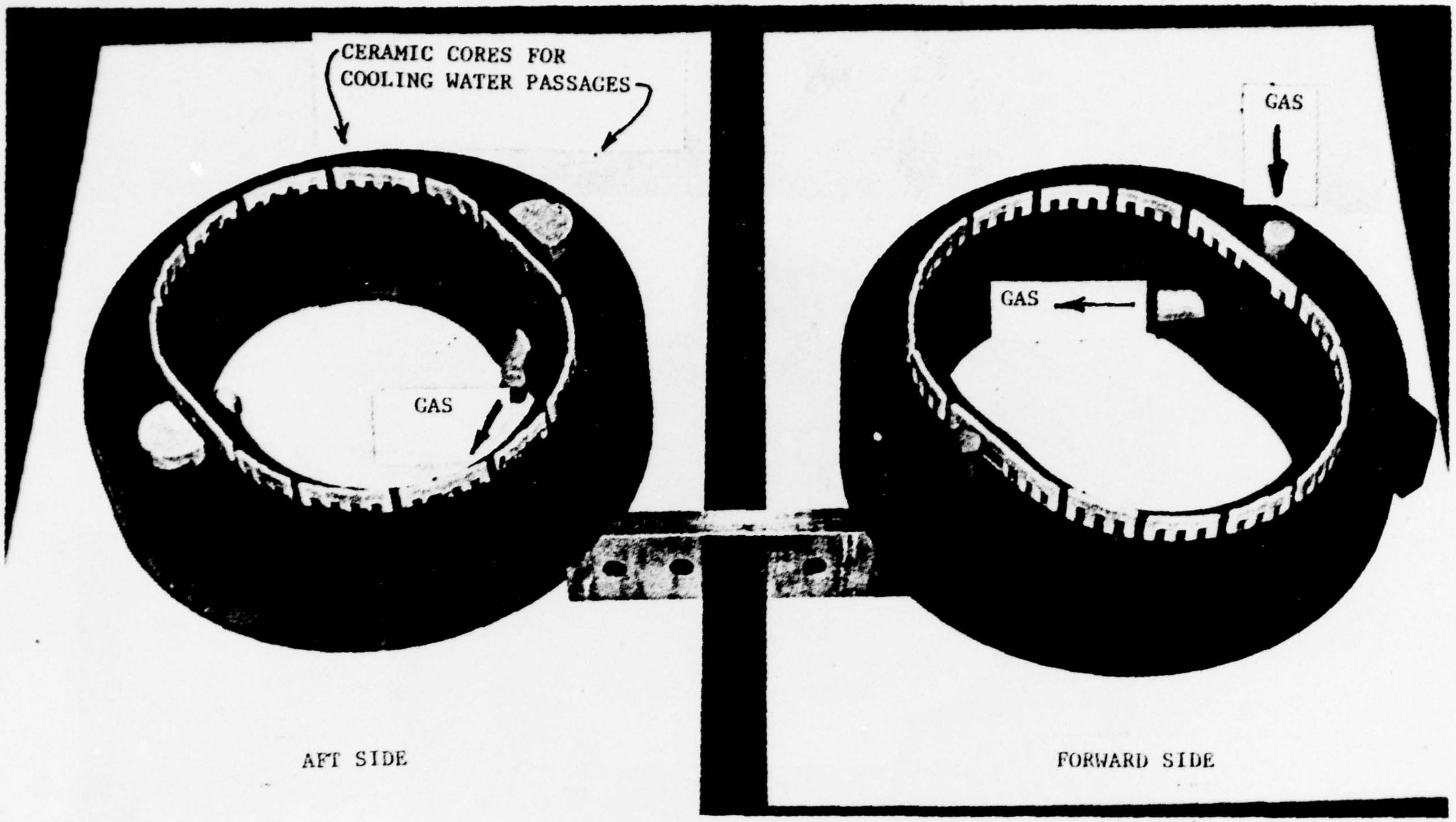


Figure 14 - Forward and Aft Side Views of Completely Assembled Rotor Housing Waxes and Cores

The completely assembled models, waxes and cores, for the forward and aft covers and rotor housing are then repeatedly dipped into a ceramic slurry which forms a shell around the wax models and simultaneously attaches itself to the ceramic cores. After sufficient shell thickness is obtained, the entire assembly is oven-fired at various heats to melt and remove the wax portion of the assembly, drive off all binders and organic impurities and yield a mold in which the original ceramic cores are now firmly attached to a new ceramic shell. The ceramic assembly is placed in core boxes and supported with sand. Molten Hastealloy C is then poured into the cavity which formerly contained the wax. After solidification occurs, the ceramic shell and cores are removed from the casting by mechanical and chemical methods. This entire process produces the rough castings shown in figure 15. The casting project was handled by Hitchiner Manufacturing Company of Milford, New Hampshire.

HALF-POWER EXPERIMENTAL EVALUATION - Initially the RE1-11 will be evaluated experimentally at a half-power level of about 100 hp (74.6 kW). It will be tested using the Naval Underwater Systems Center's Steam Generator Facility which will produce superheated steam at mass flow rates to 100 lbms/min (45.3 kgm/min) at 2000 psia (13.8 MPa) and 1300^{o}F (704^{o}C). In addition to determination of torque and speed data, all gas, cooling water and lubrication oil flow circuits will be instrumented to determine the mass flow rates, and the inlet and outlet pressures and temperatures. The most significant difference between the testing of the RE1-60 and the RE1-11 is the omission of pressure instrumentation to obtain pV diagrams for the RE1-11. The kinematic arrangement does not offer any simple method to access the working volume of the expander. The computer simulation previously described will be used with measured mass flow rate and heat-transfer rate data to determine the values of the inlet and exhaust port discharge coefficients and the heat-transfer coefficient. The indicated horsepower will be predicted for the RE1-11 and when combined with estimates of the mechanical efficiency of the expander will produce a predicted brake horsepower.

Figure 15 - Hastealloy C Castings of Forward Cover, Rotor Housing and Aft Cover

CONCLUSIONS

The concept of a rotary expander engine was demonstrated experimentally in the first phase of this development program using the RE1-60. In addition to the experimental data, analytical design techniques, based on the computer simulation of the thermodynamic, heat-transfer and fluid dynamic processes for the device, were developed. To date, the second phase of the program to produce a 250 hp (186 kW) rotary expander engine has progressed to the stage where the new expander, the RE1-11, has been designed and partially fabricated, and plans for half-power testing developed. Although the RE1-11 incorporates all the changes suggested by the RE1-60 tests, and was designed using computer techniques from the first phase of the program, the kinematic arrangement of the RE1-11 is a radical departure from that of the RE1-60, and a return to Wankel's earliest concepts. The advantages gained in this new design concept are significant, and include the elimination of balance weights and a flywheel, the location of the combustion chamber and expander working volume closer to each other, and the incorporation of highly efficient flow paths for the combustion gases, cooling water and lubrication oil.

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Mr. Richard W. Boeglin and Mr. Alfred D. Silvia made significant contributions to the design phases of this program. This research was sponsored by the Research and Technology Directorate, Naval Sea Systems Command, Washington, D. C. and was conducted by the Naval Underwater Systems Center, Newport, Rhode Island.

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Energy Conversion Branch
Weapon Systems Department
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This paper describes some results and the status of a program to develop a 250 hp(186 kW)rotary expander engine for use with an external combustion source or in a closed cycle power system. In the first phase of the program, parts from a Curtiss-Wright RC1-60 internal combustion engine were combined with a new housing, end plates and a set of rotary valves to produce the first rotary expander, the RE1-60. The RE1-60 was tested at a maximum power output of 155 hp(116 kW) at 4800 rpm when supplied with combustion gas at 1000°F(538°C) and 600 psia(4.14 MPa). Details of the RE1-60 and test results

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The second rotary expander, the RE1-11, was designed with the expander's kinematics similar to that of Wankel's earlier engines; the main shaft/eccentric is stationary while the rotor and housing rotate. Power take-off occurs from the housing. Compared to the RE1-60 this kinematic change eliminates two balance weights and a fly wheel, and permits location of the valves closer to the expander's working chamber. Details of the RE1-11 design are given in the paper along with solutions to some challenging casting and fabrication problems. The current status of the RE1-11 fabrication program and plans for half-power testing are summarized.